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NASA TECHNICAL NOTE

NASA TN D-3101

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by Erwin V. Zaretsky, Richard J. Parker, William J. Anderson, and David W. Reichard

Lewis Research Center
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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • NOVEMBER 1965

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SUMMARY

Rolling-contact fatigue tests were performed on SAE 52100 207-size deep-groove ball bearings to determine the relation between bearing fatigue life and actual bearing component hardness differences and the effect of actual component hardness differences on bearing fatigue life scatter. The 207-size bearings with inner and outer races from the same heat of SAE 52100 material and with nominal Rockwell C hardnesses of 63 were assembled with SAE 52100 balls from the same heat of material tempered to nominal Rockwell C hardnesses of 60, 63, 65, and 66. Test conditions included an inner race speed of 2750 rpm, a radial load of 1320 pounds, which produced maximum Hertz stresses of 352 000 and 336 000 psi at the inner and the outer races, respectively, and a highly purified naphthenic mineral oil as the lubricant.

Subsequent to testing, the bearings were disassembled, and all component hardnesses were measured. The bearings were regrouped according to their actual values of ΔH for Rockwell C hardness increments of 0.5 and 1.0, where ΔH is the difference between the actual hardness of the rolling elements in the bearing and the actual hardness of the inner race. The fatigue life and scatter results were compared with component hardness combinations and data previously obtained from the five-ball fatigue tester. The following results were obtained:

The bearings exhibited a maximum life for a ΔH of approximately 1 to 2 points Rockwell C. These results correlated with those obtained with the five-ball fatigue tester. For actual ΔH increments of 0.5, the maximum life was obtained for a ΔH of 1.5 to 2 points Rockwell C and is approximately four times the life for the nominal ΔH of approximately 2 points Rockwell C. These results substantiate that maximum bearing fatigue life is obtained when the rolling elements are 1 to 2 points Rockwell C harder than the races.

For both the full-scale bearings and the five-ball system, fatigue scatter decreased with increasing ΔH until a minimum value was obtained at a value of ΔH of approximately 2 points Rockwell C. Beyond this value the fatigue life scatter increased.

INTRODUCTION

In aerospace applications high reliability is of paramount importance. Of almost equal importance is the economic consideration, where equipment downtime and maintenance must be minimized. A major factor affecting equipment reliability and operating costs is the life of the rolling-element bearings used in the equipment. Unfortunately, failure of these bearings due to fatigue is to be expected. The problem is, therefore, one of increasing bearing fatigue life.

Recent developments in bearing steel alloys has provided marked increases in rolling-element fatigue life and reliability. The development of new and improved alloys, however, has advanced the state of the art to the point where further major metallurgical improvements in bearing steels are not immediately anticipated. Research must therefore concentrate on improving the physical and metallurgical properties and mechanical applications of available bearing steels.

The five-ball fatigue tester was used to study the effect of component hardness combinations on the fatigue life of rolling elements subjected to repeated stresses applied in rolling contact (ref. 1). SAE 52100 steel balls from the same heat of material were tempered to a range of Rockwell C hardnesses from 60 to 66. Groups of upper test balls with nominal Rockwell C hardnesses of 60, 62, and 65 were run against lower test balls of nominal hardnesses of 60, 62, 63, 65, and 66. These results indicated that for a specific upper test ball hardness, the rolling contact fatigue life and load-carrying capacity of the test system increased with increasing lower test ball hardness to an intermediate hardness value, where a peak life was attained. For further increases in hardness of the lower test balls, system life and capacity decreased. The peak-life - hardness combination occurred for each of the three lots of upper test balls in which the hardness of the lower test ball was approximately 1 to 2 points Rockwell C greater than that of the upper test ball. According to these results for SAE 52100 steel, a maximum bearing fatigue life should occur when the balls of the bearing are 1 to 2 points harder than the races.

Differences in plastic deformation of the rolling surfaces and contact temperature for different hardness combinations could not account for measured differences in life.

Additional work reported in reference 2 indicates an interrelation among differences in component hardness, induced compressive residual stress, and fatigue life. Examination of the data in reference 1 also indicates that there may be some interrelation between the bearing fatigue life scatter, as indicated by the slope of a Weibull plot, and component hardnesses.

Tests of 207-size deep-groove ball bearings of SAE 52100 steel, which were run at a radial load of 1320 pounds (which produced maximum Hertz stresses of 352 000 and 336 000 psi at the inner and the outer races, respectively) with a highly refined naphthenic mineral oil as the lubricant, are reported in reference 3. These bearing races had a

nominal Rockwell C hardness of 63. The balls of the bearing were divided into four groups and were tempered to produce nominal Rockwell C hardnesses of 60, 63, 65, and 66.

These tests indicated that maximum bearing fatigue life and load capacity are achieved when the rolling elements of a bearing are 1 to 2 points Rockwell C harder than the races. As a result of heat-treating irregularities, however, most bearings manufactured have a variation in component hardness of ±2 points Rockwell C for the races and ±1 point Rockwell C for the rolling elements, that is, balls and rollers. This would indicate as an example, that where there are intentions of making a bearing having balls and races of equal hardness, there is a probability of having a ΔH of ±3 Rockwell C. The objectives of the research reported herein were, therefore, to determine (1) the relation between bearing fatigue life and <u>actual</u> bearing component hardness differences and (2) the effect of <u>actual</u> component hardness differences on bearing fatigue life scatter.

In order to accomplish these objectives, the 207-size bearings tested under the conditions reported in reference 3 were disassembled, and all component hardnesses were measured. The bearings were regrouped according to their actual values of ΔH for Rockwell C hardness increments of 0.5 and 1.0, where ΔH is the difference between the actual hardness of the rolling elements in the bearing and the actual hardness of the inner race. The data were reanalyzed, and fatigue life and scatter results were compared with component hardness combinations and data obtained in the five-ball fatigue tester as reported in reference 1.

SPECIMENS AND PROCEDURE

Fatigue tests were conducted with 207-size radial ball bearings. The dimensions of the bearing were as follows:

Track diameter, in.	
Inner race	1.6648
Outer race	2.5411
Number of balls	9
Ball diameter, in.	0.4375
Conformity, percent	
Inner race	51
Outer race	52

The inner and the outer races were manufactured from the same heat of air-melted SAE 52100 steel and heat treated as follows:

(1) The races were austenitized at 1500° F for 20 minutes, 1530° F for 20 minutes, 1540° F for 20 minutes, and then oil quenched.

(2) The inner races were tempered at 320° F for one hour, air cooled, and then given a second temper at 330° F for 30 minutes.

(3) The outer races were tempered at 320° F for one hour, air cooled, and then given a second temper at 320° F for 30 minutes.

(4) The final nominal hardness of the inner and outer races was Rockwell C 63.

All the balls for the bearings were made from one heat of air-melted SAE 52100 steel and heat treated as follows:

TABLE I. - BALL TEMPERING SCHEDULE FOR SAE 52100 STEEL BALLS RUN IN 207-SIZE BEARINGS

[Ball diameter, 0.4375 in.; duration of each temper, 1 hr.]

Nominal Rockwell C hardness	Temperature, $^{\circ}$F	
	First temper	Second temper
60	250	435
63	250	330
65	250	260
66	200	None

The balls were stabilized at 1250° F for 40 minutes, austenitized at 1500° F for 30 minutes, and then oil quenched. The as-quenched hardness of the balls was approximately 66.5 Rockwell C. The balls were then tempered according to the schedule shown in table I.

The balls and the races were finished to an AFBMA 10 and an ABEC 5 specification, respectively. The bearings were assembled into four groups based on ball nominal Rockwell C hardnesses of 60, 63, 65, and 66. The bearings were tested at ambient temperature, at an inner-race speed of 2750 rpm, at a radial load of 1320 pounds, and with a highly purified naphthenic mineral oil as the lubricant.

At the conclusion of tests under these conditions, the bearings were disassembled. Parallel flats approximately 1/8 inch in diameter were ground on two balls of each bearing randomly selected to facilitate eight hardness measurements (four per ball). Four measurements were taken at regularly spaced locations on each of the bearing races. The values of these measurements were averaged and used to represent the actual hardness of each component. These average values were used to determine the actual ΔH values (actual hardness of the balls minus the actual hardness of the inner race) as distinguished from the nominal ΔH values in table II for each of the bearings. The four groups of bearings were then segregated according to tables III and IV, respectively.

Total running time for each bearing was recorded in number of inner-race revolutions. The statistical methods outlined in reference 4 were used to obtain a plot of the log log of the reciprocal of the probability of survival as a function of the log of stress cycles to failure (Weibull coordinates).

TABLE II. - BEARING FATIGUE LIFE WITH VARYING BALL HARDNESS FOR 207-SIZE DEEP-GROOVE BALL BEARINGS

[Radial load, 1320 lb; speed, 2750 rpm; race nominal Rockwell C hardness, 63.]

Nominal ball Rockwell C hardness	Nominal ΔH (difference in hardness between balls and race)	10-Percent fatigue life, millions of inner-race revolutions	Bearing load capacity based on experimental life, C, lb	Confidence number, percent[a]	Failure index (number of bearings failed out of number of bearings tested)
60	-3	21	3640	89	14 out of 28
63	0	77	5620	60	11 out of 25
65	2	106	6250	--	12 out of 28
66	3	74	5540	62	14 out of 27

[a]Percentage of time that 10-percent life obtained with each hardness combination will have same relation to hardness combination in that series exhibiting highest 10-percent life.

TABLE III. - BEARING FATIGUE LIFE WITH VARYING BALL HARDNESS FOR 207-SIZE DEEP-GROOVE BALL BEARINGS WITH ACTUAL ROCKWELL C HARDNESS INCREMENT OF 0.5

[Radial load, 1320 lb; speed, 2750 rpm.]

Actual ΔH (ball Rockwell C hardness minus inner-race Rockwell C hardness)	Weibull slope	Fatigue life, millions of inner-race revolutions		Ratio of 50-percent life to 10-percent life	Failure index (number of bearings failed out of number of bearings tested)
		10-Percent life	50-Percent life		
-1.5 to -1.0	0.9	9.0	72.0	8.0	8 out of 15
-1.0 to -0.5	1.1	34.0	200.0	5.9	3 out of 8
0.5 to 1.0	1.0	15.5	106.0	6.9	3 out of 3
1.0 to 1.5	1.1	70.0	365.0	5.2	7 out of 14
1.5 to 2.0	[a]1.7	405.0	1220.0	3.0	1 out of 8
2.0 to 2.5	1.8	92.0	265.0	2.9	9 out of 11
2.5 to 3.0	1.4	78.0	295.0	3.8	10 out of 17
3.0 to 3.5	1.1	82.0	505.0	6.2	7 out of 22
3.5 to 4.0	[b]1.0	22.0	136.0	6.1	2 out of 5

[a]Slope value assumed to be that of ΔH between 1.5 and 2.5.
[b]Slope value assumed to be that of ΔH between 3.0 and 4.0.

TABLE IV. - BEARING FATIGUE LIFE WITH VARYING BALL HARDNESS FOR 207-SIZE DEEP-GROOVE BALL BEARINGS WITH ACTUAL ROCKWELL C HARDNESS INCREMENT OF 1.0

[Radial load, 1320 lb; speed, 2750 rpm.]

Actual ΔH (ball Rockwell C hardness minus inner-race Rockwell C hardness)	Weibull slope	Fatigue life, millions of inner-race revolutions		Ratio of 50-percent life to 10-percent life	Failure index (number of bearings failed out of number of bearings tested)
		10-Percent life	50-Percent life		
-2.0 to -1.0	0.9	8.0	64.5	8.1	9 out of 16
-1.0 to 0	1.0	24.8	178.0	7.2	6 out of 11
0 to 1.0	1.0	15.2	106.0	7.0	3 out of 3
0.5 to 1.5	1.6	53.0	290.0	5.7	10 out of 17
1.0 to 2.0	1.1	108.0	605.0	5.5	8 out of 22
1.5 to 2.5	1.7	129.0	400.0	3.1	10 out of 19
2.0 to 3.0	1.7	90.0	276.0	3.1	19 out of 28
2.5 to 3.5	1.3	84.0	350.0	4.2	17 out of 39
3.0 to 4.0	1.0	64.5	435.0	6.8	9 out of 27

RESULTS AND DISCUSSION

Fatigue Results

Results of the fatigue tests of the four groups of bearings with races of nominal Rockwell C hardness 63 and balls of nominal Rockwell C hardnesses of 60, 63, 65, and 66 (ref. 3) are presented in figure 1 and summarized in table II.

The 10-percent life (life in which 90 percent of the specimens will survive) was used for comparitive purposes. The confidence numbers for the 10-percent lives were calculated and are also given in table II. These confidence numbers indicate the percentage of the time that the 10-percent life obtained with each hardness combination will have the same relation to the hardness combination exhibiting the highest 10-percent life. Thus, a confidence number of 90 means that 90 out of 100 times the specimens tested with a given hardness combination will give a life relation similar to those presented. The failure index given in table II indicates the number of failures out of the number of bearing specimens tested.

The relative bearing 10-percent lives (data from table II) are plotted as a function of ΔH, the difference in hardness between the balls and the races, in figure 2. A maximum life exists where ΔH is between 1 and 2 points Rockwell C.

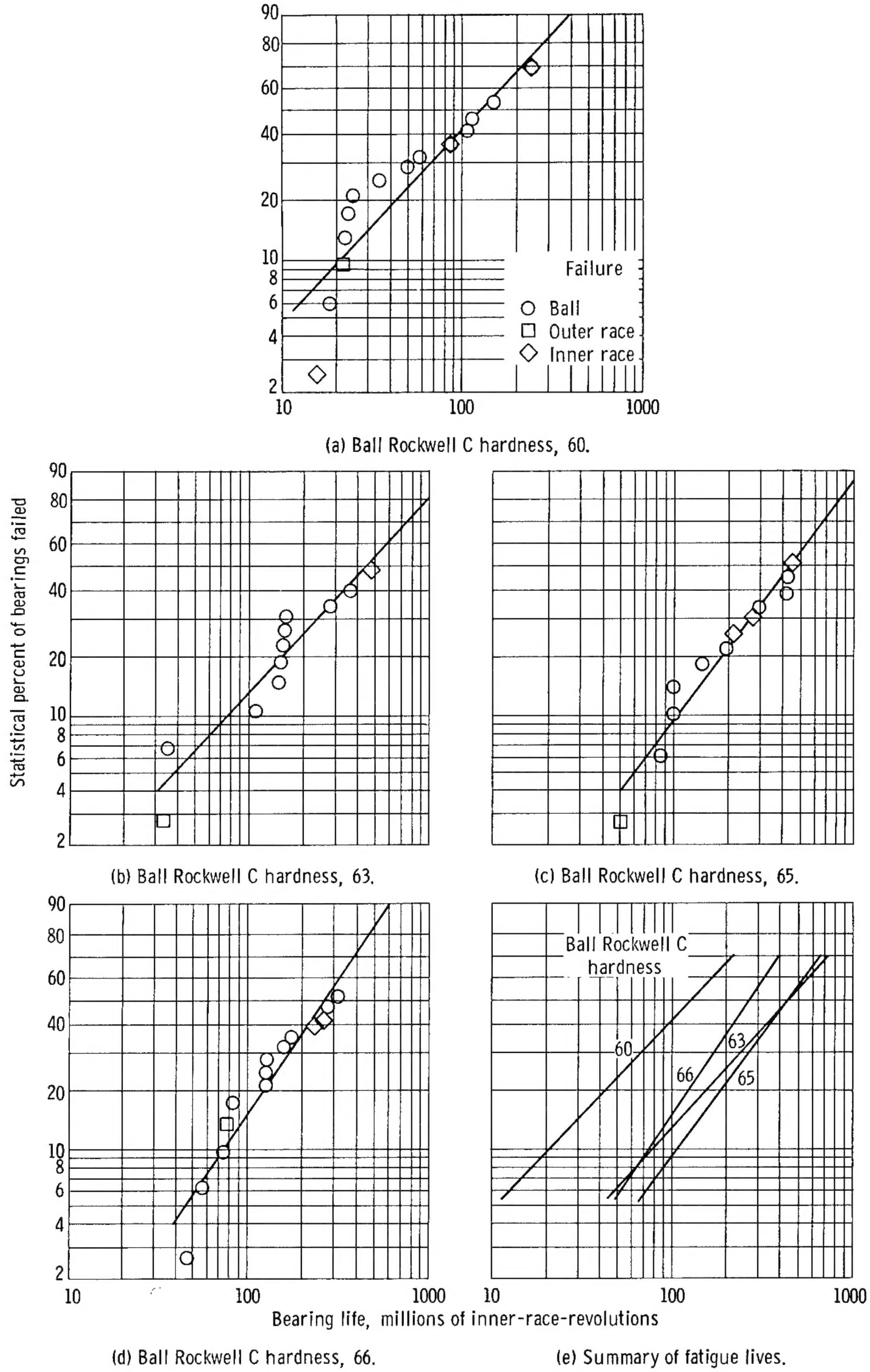


Figure 1. - Rolling-contact fatigue life of 207-size deep-groove ball bearings with SAE 52100 races of nominal Rockwell C hardness of 63 and SAE 52100 balls of varying hardness. Radial load, 1320 pounds; speed, 2750 rpm; no heat added.

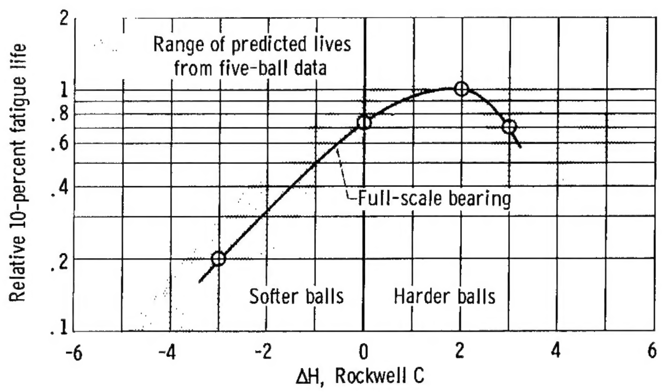


Figure 2. - Relative 10-percent life as function of ΔH (difference in Rockwell C hardness between balls and races) for 207-size deep-groove ball bearings with SAE 52100 races of nominal Rockwell C hardness 63 and SAE 52100 balls of varying hardnesses. Radial load, 1320 pounds; speed, 2750 rpm; no heat added.

An important criterion of bearing operation is the load-carrying capacity. This capacity is the load that the bearing can sustain for 1 million inner-race revolutions with a 90-percent probability of survival. The load-carrying capacity of each of the bearings tested may be calculated from the fatigue data results summarized in table II (p. 5) by use of the following equation:

$$C = P \sqrt[3]{L}$$

where

C load capacity, lb

P load on bearing, lb

L 10-percent life, millions of inner-race revolutions

The calculated capacities for the bearings are given in table II. As with the fatigue results, a maximum capacity exists at a value of ΔH of 1 to 2 points Rockwell C.

The five-ball fatigue data from reference 1, obtained under test conditions similar to those for the bearings but at a maximum Hertz stress of 800 000 psi are summarized in table V. These five-ball system 10-percent lives are plotted as a function of the bearing 10-percent lives in figure 3 for the same hardness combinations, that is, upper-ball (inner-race) hardness of 63 Rockwell C and lower-ball (rolling-element) nominal hardnesses of 60, 63, 65, and 66 Rockwell C. These data indicate a correlation between the five-ball system and full-scale bearings, even though the five-ball-system data were obtained at a higher Hertz stress. From the five-ball data (table V) a range of relative five-ball-system lives are presented as the shaded area in figure 2. As was expected, the relative 10-percent fatigue lives of the bearings fell within the relative life range of the five-ball system for equivalent values of ΔH.

Effect of Component Hardness Quality Control on Fatigue Life and Scatter

As a result of heat-treating irregularities, most bearings manufactured, in general, have a variation in component hardness of ±2 Rockwell C for races and ±1 Rockwell C for rolling elements, that is, for balls and rollers. This variation would indicate that where there is an intention of making a bearing with balls and races of equal hardness, there is

TABLE V. - SYSTEM FATIGUE LIFE AND LOAD CAPACITY

OBTAINED WITH VARYING HARDNESS COMBINATIONS

IN FIVE-BALL FATIGUE TESTER

[Initial maximum Hertz stress, 800 000 psi; system thrust load, 340 lb; contact angle, 30°; room temperature; material, SAE 52100 steel; data from ref. 1.]

Upper test ball Rockwell C hardness	Lower test ball Rockwell C hardness	Difference in Rockwell C hardness between lower and upper test balls, ΔH	10-Percent fatigue life, millions of upper-ball revolutions	Weibull slope
60.5	60.5	0	2.1	0.9
	61.9	1.4	5.8	1.4
	63.2	2.7	2.7	.8
	65.2	4.7	3.2	.8
	66.4	5.9	3.3	1.0
63.2	59.7	-3.5	0.4	0.9
	61.8	-1.4	1.4	1.1
	63.4	.2	2.6	1.1
	65.0	1.8	4.7	1.3
	66.2	3.0	1.2	.8
65.2	60.5	-4.7	0.6	0.8
	61.9	-3.3	1.2	1.2
	63.2	-2.0	1.7	1.1
	65.2	0	3.9	1.2
	66.4	1.2	5.7	1.3

a probability of having a ΔH of ± 3 Rockwell C. From the data presented in table II, a ΔH of -3 points Rockwell C can account for more than a 70 percent reduction in fatigue life when compared with a ΔH of 0. It was therefore theorized that if the value of ΔH in any particular group of bearings were closely controlled, bearing fatigue life scatter would decrease. It was further theorized that the life of a bearing with a closely controlled value of ΔH would greatly exceed the maximum life indicated by the bearing data presented in figure 1 (p. 7) and table II (p. 5). Therefore, to determine the effect of quality control on bearing fatigue life as a function of ΔH and to determine whether the variation in component hardness can affect bearing fatigue life scatter, each of the 207-size bearings was disassembled, and each component was measured as previously discussed.

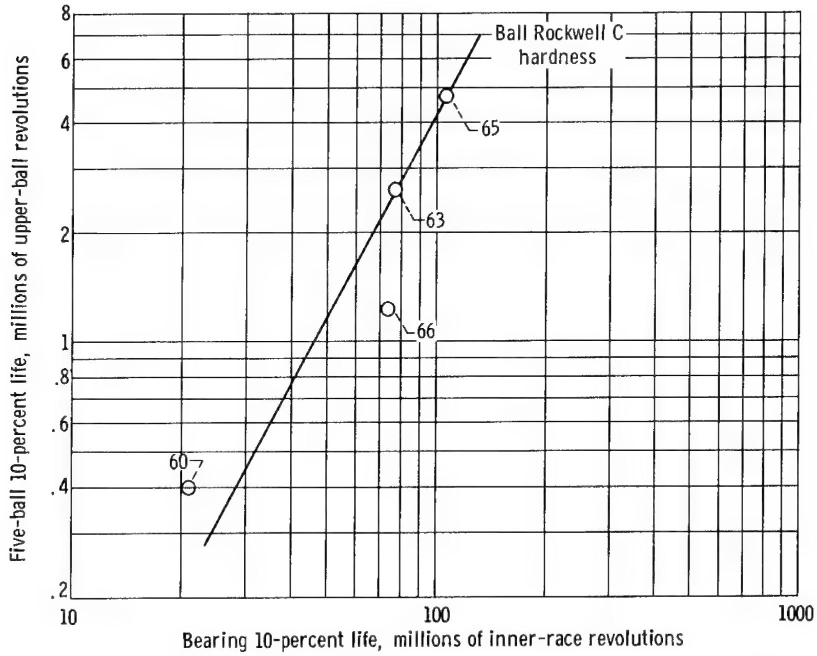
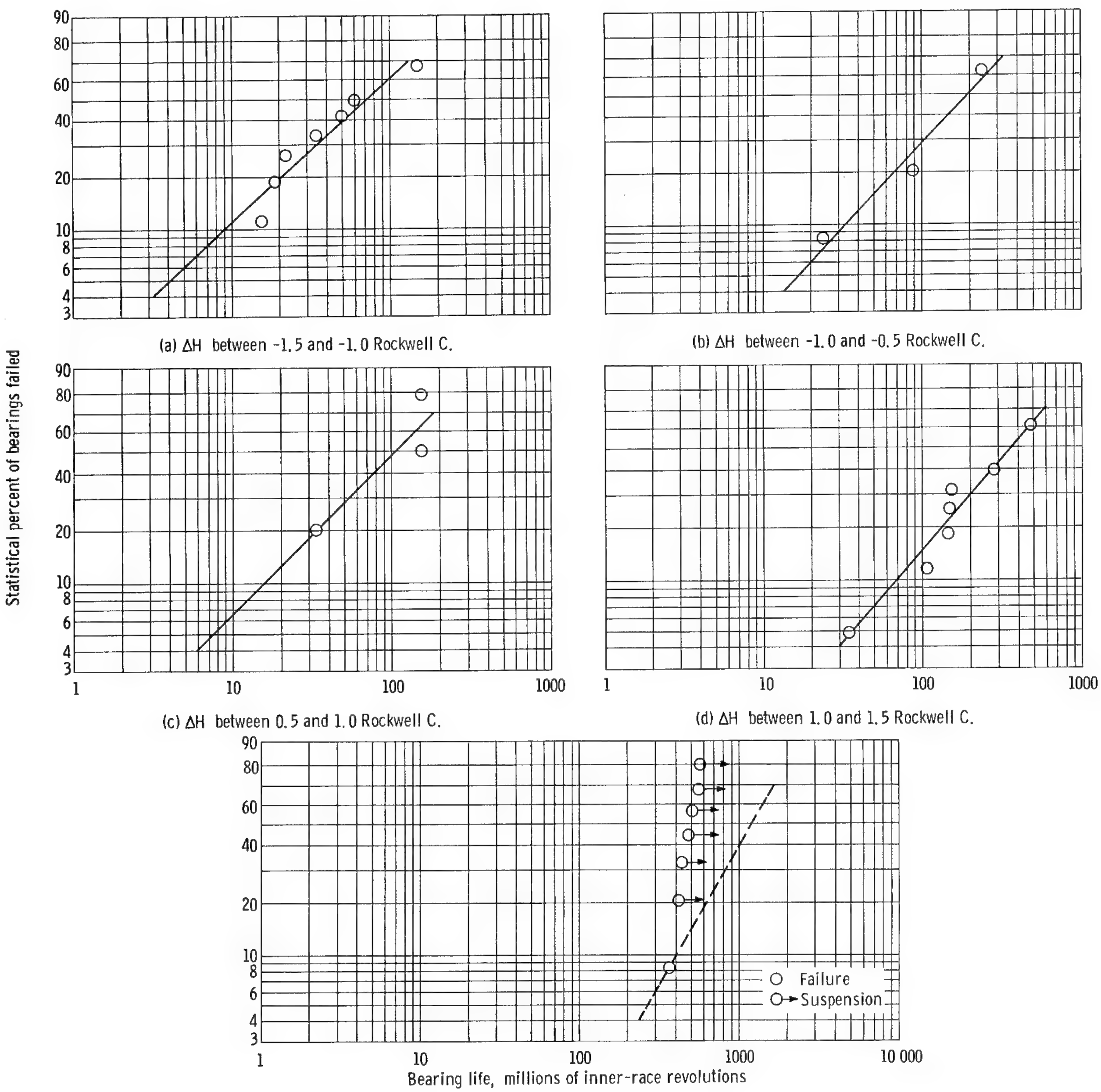


Figure 3. - Ten-percent life of five-ball fatigue system as function of 10-percent life of 207-size bearings. Inner race and upper test ball Rockwell C hardness, 63; material, SAE 52100 steel, run with SAE 52100 balls of varying hardnesses.

The life results for actual ΔH increments of 0.5 are presented in figure 4 and are summarized in table III (p. 5); the data for actual ΔH increments of 1.0 are presented in figure 5 and are summarized in table IV (p. 6). The 10-percent lives for ΔH increments of 0.5 and 1.0 are summarized in figure 6. The peak life (405 million revolutions) for ΔH increments of 0.5 occurs at an actual ΔH between 1.5 and 2.0 points Rockwell C (fig. 6(a)). Additionally, this peak life is approximately four times the peak life (106 million revolutions) obtained at a nominal ΔH of 2 for the initial bearing data presented in table II (p. 5). For actual ΔH increments of 1.0 (fig. 6(b)) the peak life is also increased but by a small amount relative to the nominal ΔH measurements (129 million revolutions compared with 106 million revolutions).

The data presented in figure 6(b) indicate that the bearings with an actual ΔH of between 1 and 2 points Rockwell C demonstrated a potential of four to five times greater fatigue life than could be achieved with bearings manufactured by normal methods.

The Weibull slopes summarized in tables III and IV for actual ΔH increments of 0.5 and 1.0, respectively, are plotted in figure 7. (The Weibull slope is a measure of the amount of scatter, that is, the greater the slope, the less the scatter. A slope of 1.1 is common for most bearing fatigue data.) Weibull slopes for the five-ball system data (table V) are plotted in figure 8. The upper test balls of the five-ball system (analogous to the inner race of the bearing) were of nominal Rockwell C hardnesses of 60, 63, and 65 and were run against lower test balls of varying hardnesses. Both the bearing and the



(a) ΔH between -1.5 and -1.0 Rockwell C.

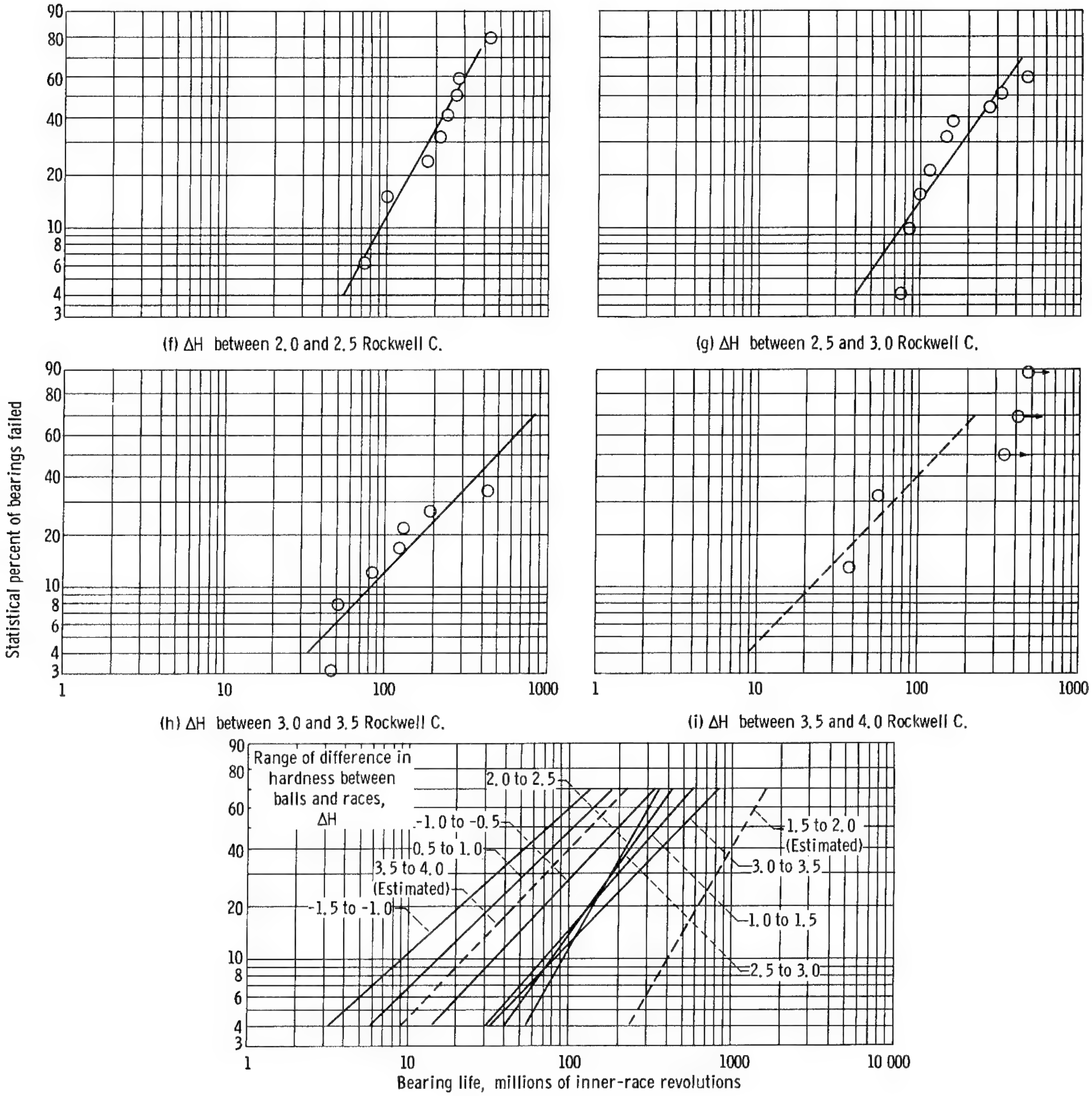
(b) ΔH between -1.0 and -0.5 Rockwell C.

(c) ΔH between 0.5 and 1.0 Rockwell C.

(d) ΔH between 1.0 and 1.5 Rockwell C.

(e) ΔH between 1.5 and 2.0 Rockwell C.

Figure 4. - Rolling-contact fatigue life of 207-size deep-groove ball bearings for actual ΔH (difference in Rockwell C hardness between balls and races) increments of 0.5. Material, SAE 52100 steel for ball and races; radial load, 1320 pounds; speed, 2750 rpm; no heat added.



(f) ΔH between 2.0 and 2.5 Rockwell C.

(g) ΔH between 2.5 and 3.0 Rockwell C.

(h) ΔH between 3.0 and 3.5 Rockwell C.

(i) ΔH between 3.5 and 4.0 Rockwell C.

(j) Summary of fatigue lives.

Figure 4. - Concluded.

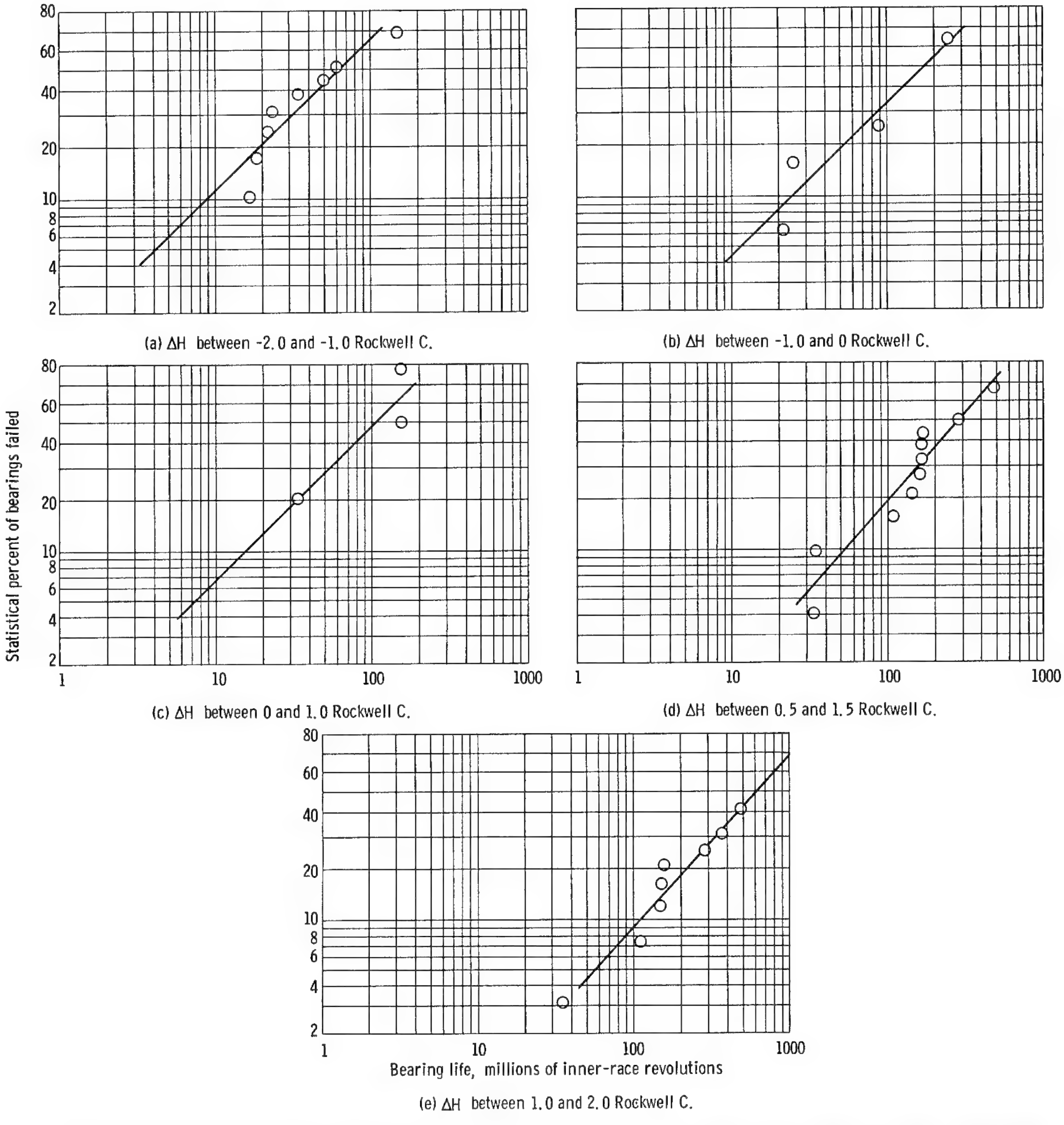
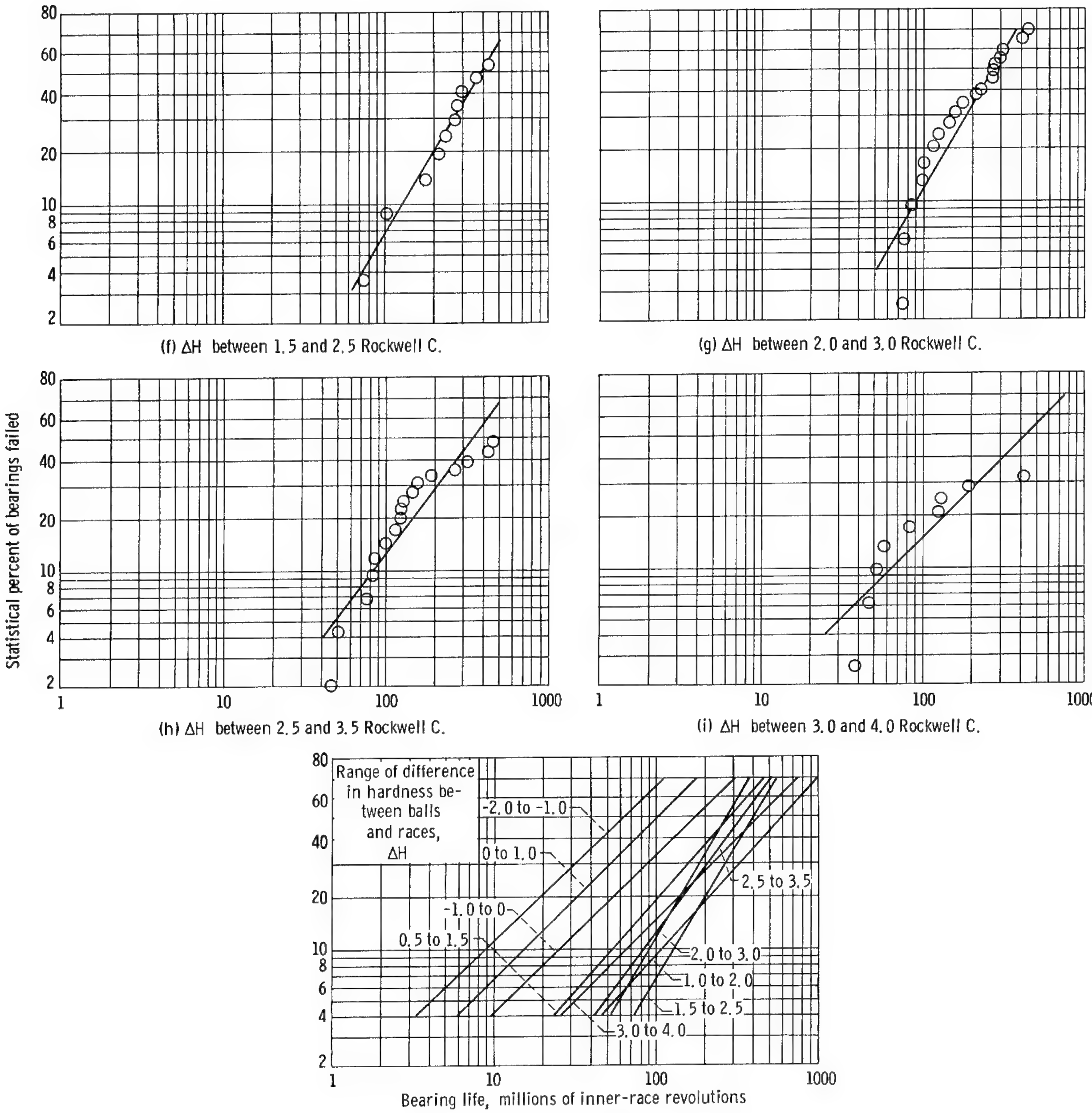


Figure 5. - Rolling-contact fatigue life of 207-size deep-groove ball bearings for actual ΔH (difference in Rockwell C hardness between balls and races) increments of 1.0. Material, SAE 52100 steel for balls and races; radial load, 1320 pounds; speed, 2750 rpm; no heat added.



(f) ΔH between 1.5 and 2.5 Rockwell C.

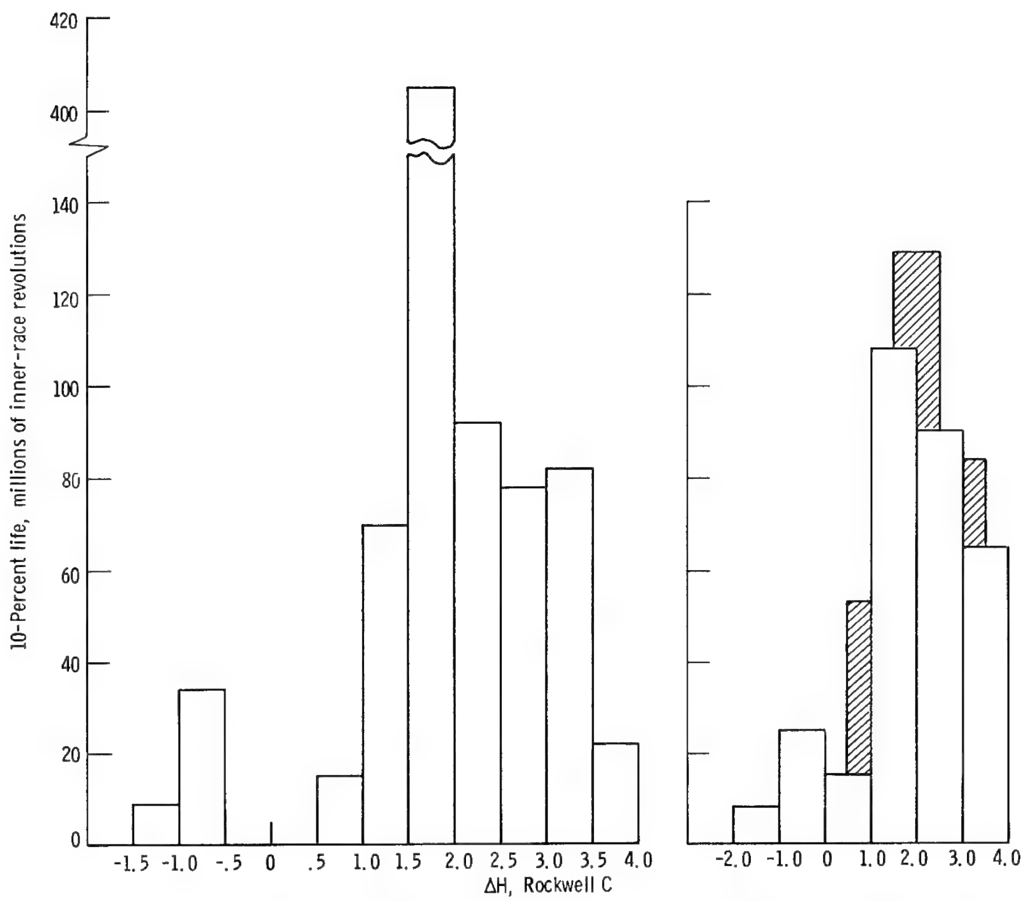
(g) ΔH between 2.0 and 3.0 Rockwell C.

(h) ΔH between 2.5 and 3.5 Rockwell C.

(i) ΔH between 3.0 and 4.0 Rockwell C.

(j) Summary of fatigue lives.

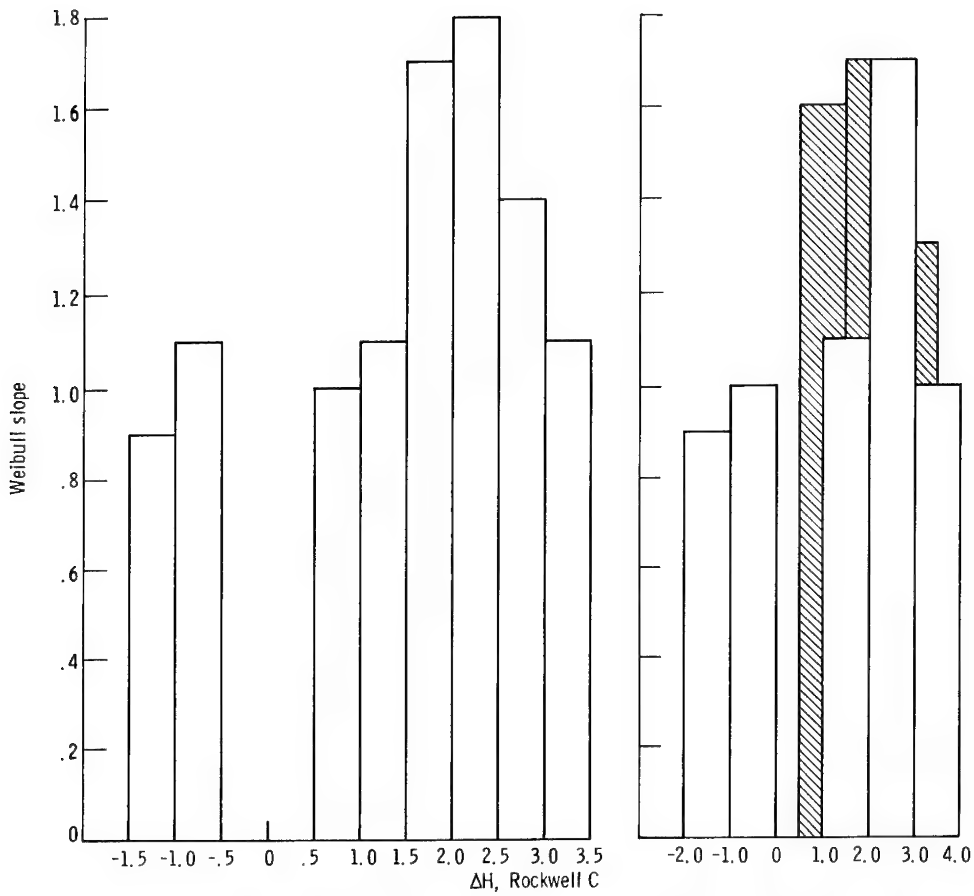
Figure 5. - Concluded.



(a) Actual ΔH increment, 0.5.

(b) Actual ΔH increment, 1.0.

Figure 6. - Ten-percent life as function of ΔH (difference in Rockwell C hardness between balls and races) for 207-size deep-groove ball bearings. Material, SAE 52100 steel for balls and races; radial load, 1320 pounds; speed 2750 rpm; no heat added.



(a) Actual ΔH increment, 0.5.

(b) Actual ΔH increment, 1.0.

Figure 7. - Weibull slope as function of ΔH (difference in Rockwell C hardness between balls and races) for 207-size deep-groove ball bearings. Material, SAE 52100 steel for balls and races; radial load, 1320 pounds; speed, 2750 rpm; no heat added.

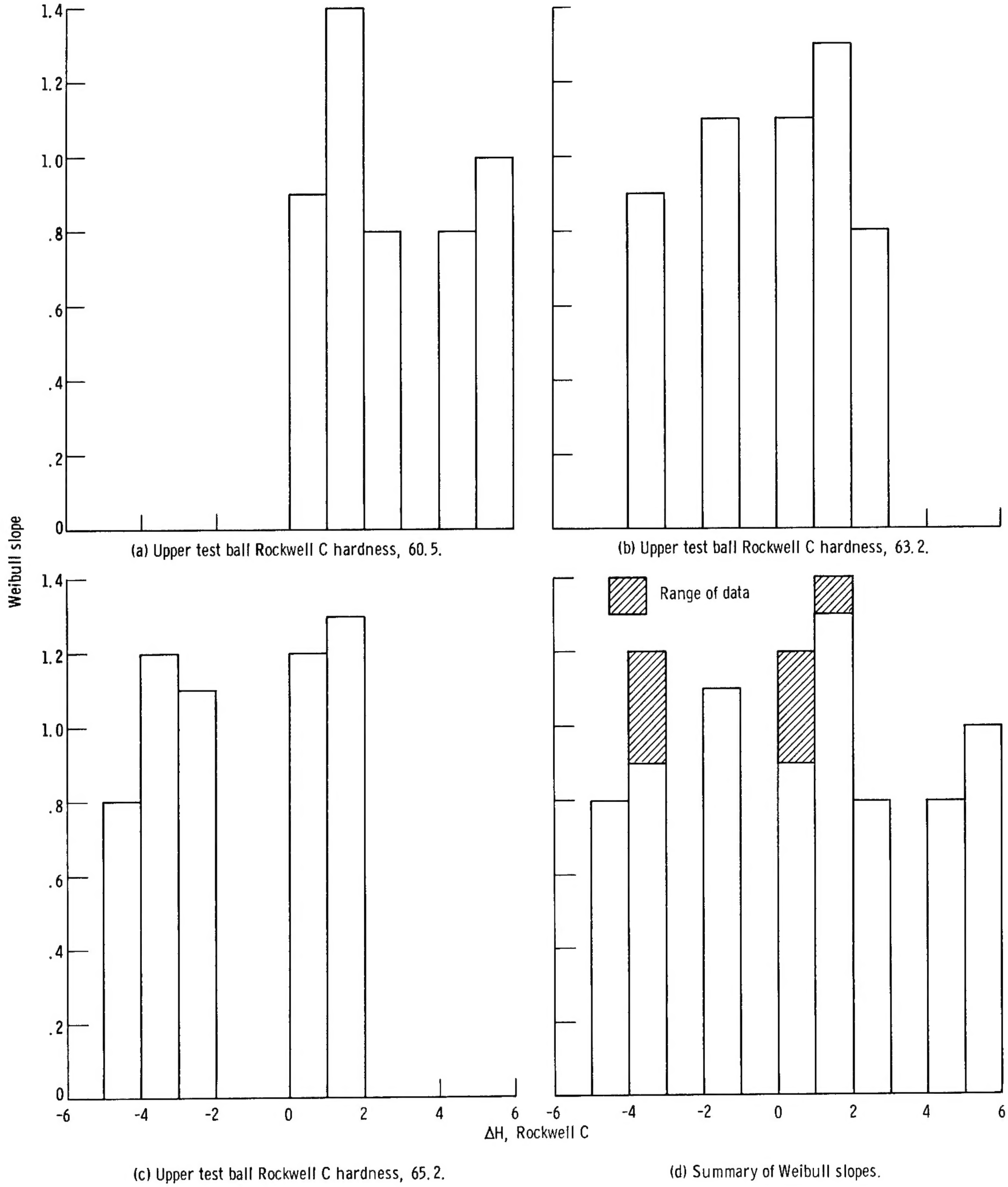


Figure 8. - Weibull slope as function of ΔH (difference in Rockwell C hardness between lower test balls and upper test ball) for upper and lower test balls of varying hardnesses. Five-ball fatigue tester; initial maximum Hertz stress, 800 000 psi; system thrust load, 390 pounds; contact angle, 30°; room temperature; material, SAE 52100 steel (data from ref. 1).

five-ball system data indicate that the amount of fatigue scatter is reduced as ΔH increases to a value of approximately 2 points Rockwell C. At this ΔH value the slopes were 1.4 and 1.8 for the five-ball system and the bearings, respectively.

From these data it can be concluded that bearing fatigue life scatter is influenced by component hardness differences. These data indicate that if the optimum ΔH is properly selected, the probability of early failures in a group can be lessened.

SUMMARY OF RESULTS

Rolling-contact fatigue tests on 207-size deep-groove ball bearings were analyzed to determine the relation between actual component hardness differences on bearing fatigue life and scatter. These tests were conducted at a radial load of 1320 pounds, which produced maximum Hertz stresses of 352 000 and 336 000 psi at the inner and the outer races, respectively, at an inner-race speed of 2750 rpm, and with a highly purified naphthenic oil as the lubricant. The following results were obtained:

1. For actual ΔH increments of 0.5 point Rockwell C, the maximum fatigue life was obtained for a ΔH range of 1.5 to 2.0 points Rockwell C where ΔH is the difference between the actual hardness of the rolling elements in the bearing and the actual hardness of the race.

2. A relation is indicated between bearing fatigue life scatter and component hardness combinations. For both the full-scale bearings and the five-ball system, fatigue scatter decreased with increasing ΔH until a minimum value was obtained at a ΔH of approximately 2 points Rockwell C.

3. The bearings having a ΔH of between 1 and 2 points Rockwell C demonstrated a potential of four to five times greater fatigue life than could be achieved with bearings manufactured with normal methods.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, September 8, 1965.

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